case being examined $E_T = 0.845$, $H = 285$ ft, and $E_P = 0.88$. Substituting these values in the foregoing equations

$$0.845 = M\,\frac{285 - L_T}{285} \quad \text{and} \quad 0.880 = M\,\frac{285}{285 + L_P}$$

or

$$L_T = 285 - \frac{240.82}{M} \quad \text{and} \quad L_P = 323.90\,M - 285$$

The following table may then be constructed:

M	1	0.98	0.96	0.94	0.92	0.90	0.88
L_T ft	44.2	39.3	34.2	28.8	23.2	17.4	11.4
L_P ft	38.9	32.4	25.9	19.4	12.9	6.5	0
$(L_T - L_P)$ ft	5.3	6.9	8.3	9.4	10.3	10.9	11.4

Evidently M exceeds 88 per cent and in all probability is over 90 per cent. The difference $(L_T - L_P)$ must be due to shock, since the water velocities are the same in both cases; it would appear that the shock losses should also be similar in the two cases. It would tend to confirm the generally accepted view that the water passages in the impeller are not completely filled, so that the distributor setting for maximum efficiency must be changed when the flow is reversed.

At this same unit speed of $N_1 = 118.5$, with $Q = 8.5$ cfs, the velocity in the $7^3/_4$-in. suction pipe is 25.9 fps, with a corresponding velocity head of 10.4 ft, which is high for such a unit. However, the writer checked out the conditions at the rim of the impeller and, taking the distributor-guide angle of 9 deg stated in the paper and the impeller discharge angle as 30 deg also as given in the paper, found that these corresponded to best efficiency, assuming that the metal in the impeller vanes occupies 11.5 per cent of the total area, which seems reasonable.

The question of cavitation in such a unit is of great importance, and serious trouble has been experienced in some of the plants in operation. Hence, a valuable addition to the paper would be the results of further study on this phase of the work. Attention might also be called to the fact that this dual-purpose unit is confined to the case where a single-stage pump suffices, because the multistage turbine has not, so far, proved attractive.

The writer raises a mild protest against the practice of stating specific speeds of pumps in terms of gallons per minute, instead of cubic feet per second. The latter is the logical unit and makes it easier to compare results of machines in general.

R. L. Daugherty.[6] This paper is a clear presentation of the characteristics of a centrifugal pump when equipped with movable diffuser vanes; and it also shows the performance of the same pump when used as a reaction turbine. There are certain cases where such dual operation may be very desirable.

In Fig. 9, the authors show the efficiency curve for this pump as obtained at the California Institute of Technology with the pump being operated at 2000 rpm. They also show an efficiency curve obtained by them in the Newport News laboratory with a pump speed of 1000 rpm, but with the test results stepped up to 2000 rpm. The agreement between the two curves is very close. It is not to be expected that the two curves should coincide.

In general, the efficiency of any centrifugal pump will increase slightly with an increase in speed, until incipient cavitation or some other factor causes it to begin to decline with further increase in speed. This is shown in Fig. 13 of this discussion, in which the writer has plotted maximum efficiencies as a function of speed for four stock centrifugal pumps which were tested at the California Institute of Technology. An inspection of this figure will show that, for an increase in pump speed from 1000 rpm to

[6] Professor of Mechanical Engineering, California Institute of Technology, Pasadena, Calif. Former vice-president, and Fellow A.S.M.E.

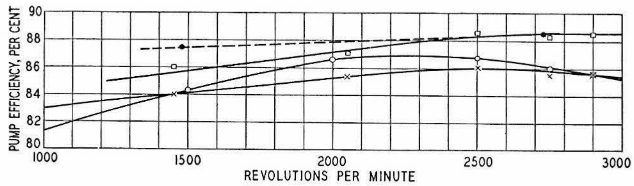

Fig. 13 Variation of Maximum Efficiencies of Four Centrifugal Pumps With Speed

2000 rpm, the difference of about 1 per cent in efficiency, shown by Fig. 9 of the paper, is entirely reasonable.

The authors state that the difference in the point where the break occurs in the head curve is caused by the fact that the intake pipes in the two laboratories were not of the same size. The method of computing the head would seem to make the size of the intake pipe of no importance, but what will influence the break in the head curve is the beginning of cavitation. Cavitation is a function of a factor defined by the equation

$$\sigma = \frac{b - p - h_f - L}{H}$$

where b is the barometric pressure, p is the vapor pressure, h_f is the friction loss in the intake pipe, L is the static suction lift, and H is the total head developed by the pump per stage. If the water is supplied to the pump by gravity then the sign of L would be positive. Cavitation begins at the same value of σ regardless of other external conditions.

A. Hollander.[7] It is only a short time ago since it was realized that efficient pumps may serve also as efficient turbines. The historical review is given in Professor Knapp's paper,[8] which also first presented the circle diagram suggested by Prof. Th. von Kármán. This diagram, with ordinates of speed and flow rate in the positive and negative directions, and constant head and torque lines, covers all possible operating conditions of the machine which the authors call the pump-turbine, including, besides the normal pump and turbine operation, the abnormal pump and turbine operation when the units are running in the direction opposite to the normal, and showing besides these four sectors of energy dissipation. Consideration of this pump-turbine as a machine equally adapted for both purposes, and going over the full circle with all possible ways of its operation, would give a better and more complete picture of its nature. In colleges, the deduction of the energy formulas and the description of velocity diagrams, on this general basis, would constitute an advance compared with present teaching methods.

It was Dr. Knapp's paper, showing for the first time these results with good pumps of about 4-in. size and 80 per cent efficiency which induced the Metropolitan Water District of Southern California to install a modern hydraulic laboratory at the California Institute of Technology,[9] in order to establish specifications and later conduct the model tests for the large pumps of 4000 to 12,000 hp, required for the Metropolitan Aqueduct of Southern California.[10]

[7] Consulting Engineer, Byron Jackson Company, Los Angeles, Calif. Mem. A.S.M.E.

[8] "Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior," by Robert T. Knapp, Preprinted Papers, A.S.M.E., Aeronautic and Hydraulic Divisions Summer Meeting, June, 1934, pp. 60–64; available at Engineering Societies' Library as A.S.M.E. Miscellaneous paper 1-N, 1934.

[9] "The Hydraulic Machinery Laboratory at the California Institute of Technology," by R. T. Knapp, Trans. A.S.M.E., vol. 58, 1936, pp. 663–676.

[10] "Centrifugal Pumps for the Colorado River Aqueduct," by R. L. Daugherty, *Mechanical Engineering*, vol. 60, 1938, pp. 295–299.

The Reclamation Bureau used the results of this laboratory with 8-in. pumps of 300 to 500 hp and continued the work for the proposed 60,000-hp pumps for Grand Coulee. The results published by Dr. Knapp (12) are further elaborated and extended by the authors. The authors are to be congratulated that they gave not only performances but most of the physical dimensions. The interest of the turbine builders in these pumps is fully justified because of the large size of some of these units, particularly for hydroelectric-storage developments. Such pumps are more in line with the turbine manufacturers' facilities than with those of most pump builders. The developments, based mainly on the California Institute of Technology tests, will certainly permit us to make our hydroelectric storage projects a great deal simpler and less costly than those in Germany, which in a number of cases could have used the same machine as pump and turbine instead of having independent units with a great number of auxiliary connections, clutching and declutching apparatus, etc.

We missed the list of the disadvantages of the wicket-gate pump-turbine, as compared with the simple volute pump-turbine, without a wicket gate, and possibly a valve to regulate flow. Undoubtedly, for some purposes, particularly for widely varying flows, the wicket-gate pump-turbine has some justification. On the other hand, such units and especially their volute cases, due to the increase in diameter by the diffuser, are of very much greater over-all dimensions than plain single or double volutes. Thus, in the end probably they will be a great deal more expensive than volute-case pumps. For applications where the levels do not change very much, it seems that the plain volute will have its place even for hydroelectric storage units and, undoubtedly, in most cases where pumping is the only application.

It seems that, for the discussion of the Grand Coulee units, members of the staff of the California Institute of Technology, which conducted all the tests of the different types and makes of models, are best qualified; having all the test results at their disposal their contribution would complete and round out the very instructive paper of the authors.

R. T. Knapp[11] and J. W. Daily.[12] The following discussion is based primarily upon the results of the test program carried out in the hydraulic-machinery laboratory of the California Institute of Technology to investigate the pumping problems involved in the Grand Coulee installation for the U. S. Bureau of Reclamation. This program was conducted under the immediate direction of Profs. T. H. von Kármán, R. L. Daugherty, and R. T. Knapp, with J. W. Daily in charge of the technical staff.

The information contained in this paper makes it possible to complete the set of comparative studies previously presented by R. T. Knapp (12) in which the performance characteristics of single-volute, double-volute, and fixed-vane diffuser pumps were compared and discussed. The pumps involved were units purchased by the hydraulic-machinery laboratory for the study referred to. The performance of the wicket-gate pump, described by the authors, was not included in this comparison because, although it was designed to meet the same specifications, it was not a part of the test program. However, during the program it was tested for the Newport News Shipbuilding and Dry Dock Company, under special arrangements made with the Bureau of Reclamation. Therefore, the writers did not feel free to use the performance of this pump at the time the former paper was prepared.

This group of pumps presents a unique opportunity for a comparison of the performance characteristics of these different types o ʃcasings, since all of the units were designed for the same head and capacity and the same range of inlet heads. Two series of units were procured, one having a specific speed corresponding to 150-rpm operation of the Grand Coulee units, and the other having a specific speed corresponding to 180-rpm operation. The wicket-gate pump described in the present paper belongs to the first series. These test units represent the most modern design practice of some of the leading hydraulic-machinery manufacturers in the country. They are comparatively large machines, having discharge nozzles of 8 in. diam and requiring over 300 hp to drive them at normal capacity. It is seen that these units are considerably larger than the average commercial pump. Therefore, it is felt that the conclusions which can be drawn from the comparison of the performance of the different units must carry considerable weight.

Before the specific comparisons of the performance of the wicket-gate pump with the other test units are presented, the writers have a few miscellaneous comments on points brought out in the paper.

1 In describing the range of conditions under which it is expected that the Grand Coulee pumps will operate, the authors state that, at the maximum dynamic head of 367 ft, the discharge will be approximately 1000 cfs. Actually, this figure is best considered to be a minimum acceptable value. In the Grand Coulee operation the more water that can be pumped at maximum head the better, since this condition occurs at the beginning of the irrigation season. Thus, the capacity at the maximum head is limited only by the obtainable steepness of the pump characteristic.

2 In the description of the tests at the California Institute of Technology, the statement was made that turbine tests were limited to a maximum of 1800 rpm for units of this size by the laboratory equipment. Actually, the limitation is one of capacity and not of speed, since speeds of better than 4000 rpm are well within the range of the test equipment.

3 In discussing Fig. 7, the authors state that for an envelope head curve the highest head occurs at zero discharge. This must be thought of as a statement of idealized conditions, because in the actual test some droop of the head curve was always encountered as zero discharge was approached due to the impracticability of obtaining complete closure of the gates.

4 In discussing Fig. 8, as compared to Fig. 7, it is concluded that a full gate as a turbine is larger than a full gate as a pump, i.e., gate No. 10 versus gate No. 8. As the writers remember the situation, preliminary tests of the pump show that, for gates Nos. 9 and 10, the maximum efficiency was lower than for gates Nos. 7 and 8, and therefore the final test gates Nos. 9 and 10 were not run. In the turbine test a run was taken at gate No. 10, but as will be noted the maximum efficiency is again lower than for gates Nos. 7 and 8. Therefore, the writers have difficulty in understanding the conclusions of the authors.

5 In discussing these same two figures, it is pointed out that the best unit speed for the machine operating as a turbine is lower than the best unit speed as a pump. It is concluded from this that "the use of a combination machine at constant speed and head would involve some sacrifice in efficiency." It is felt by the writers that this generalization is not justified. The statement is certainly true as applied to the specific unit. However, if a different design point were used it would be possible to obtain equal efficiencies for both modes of operation. For example, this point is excellently illustrated by use of Fig. 14 of this discussion, which is a reproduction of Fig. 15 of the paper (12) previously mentioned. It will be seen from this diagram that a turbine operating at 100 per cent speed and normal head would have an efficiency about 3 per cent lower than the same unit operating as a pump at the same head and speed. However, if operating conditions are chosen with normal head and 96 per cent normal speed

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[12] Instructor, Mechanical Engineering, California Institute of Technology. Jun. A.S.M.E.

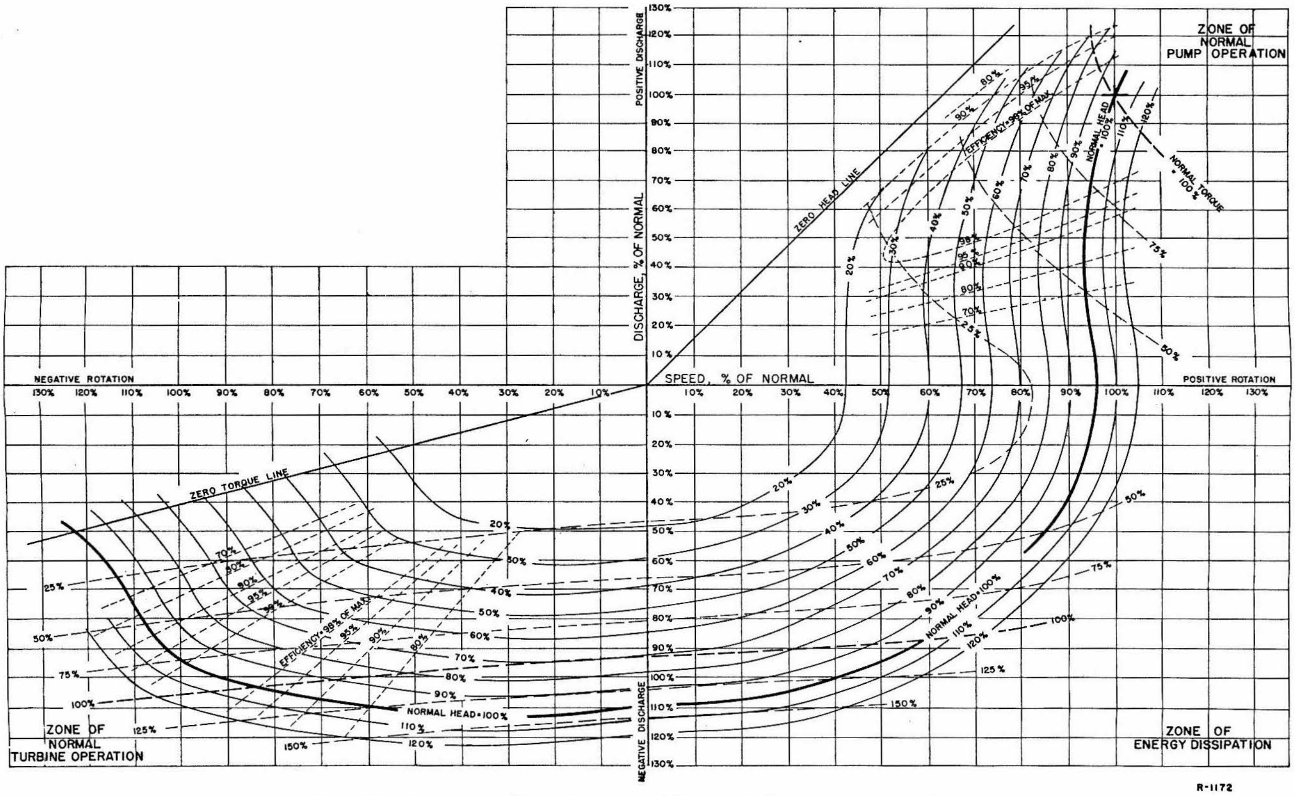

FIG. 14 COMPLETE CHARACTERISTIC DIAGRAM, DOUBLE-VOLUTE PUMP
(Reproduced from R. T. Knapp paper, authors' Bibliography 12, Fig. 15.)

both as a pump and a turbine, it will be seen that the maximum possible efficiency is obtained from both units. On the other hand, if operating conditions as pump and turbine had been chosen as normal head and 110 per cent normal speed, the diagram shows that the pump would still be operating within 3 per cent of its maximum efficiency, whereas the turbines would have dropped off about 25 per cent.

6 In presenting some of the general results of the California tests, the authors called attention to the fact that the efficiency as either a pump or a turbine was not affected until the impeller was cut below 13 in. in diameter, whereas cutting the discharge tips of the impeller vanes while leaving the shrouds extended lowered the efficiency. The writers feel that this test result has interesting implications, i.e., that the close clearance between the runner and the gate vanes generally incorporated in turbine design may not be necessary, and that there might be advantage in the pump practice of using large axial and radial clearances at the periphery of the wheel.

7 In comparing the Newport News and California tests, as illustrated in Fig. 9, the authors explain the smaller discharge at which the break in the head curve occurs in the California test by the small size of the suction pipe used. They say "the entrance loss and friction loss in this pipe were the factors which determined the cutoff point in the California test." However, in the California tests, the total dynamic inlet head was measured at a point only one diameter upstream from the impeller eye, so that the effect of any inlet-piping loss on the calculated inlet or net head produced was eliminated. It is probable that the discrepancy observed in the shutoff point is due to cavitation caused by a difference in the relative submergence at which the two tests were run. The California tests were made with an 80-ft submergence. This corresponds to a lift of 4.7 ft under the conditions of the test at Newport News. If the Newport News tests were run with an inlet pressure greater than this, the cutoff point would be expected to occur at a correspondingly higher capacity.

8 In preparing Fig. 11, the authors apparently stepped up the model efficiency to the expected prototype values by the use of the normal turbine step-up formulas. Thus the maximum model efficiency of 87.6 per cent has been increased to an expected prototype efficiency of 92 per cent. At this point it should be noted that it is the practice of the hydraulic-machinery laboratory of the California Institute of Technology to test the units at full prototype heads and velocities. In the case of the present units, this means that the Reynolds number of the flow in the suction and discharge nozzles is in the neighborhood of 2,000,000. Thus, comparatively little change in relative losses can be expected for any possible increase in the size of the units. This conclusion is borne out by the few comparisons between model and prototype test efficiencies in which models were tested at full prototype heads and velocities. Of the $4^1/_2$ per cent step-up assumed, 1 per cent is justified, as the authors point out, by the excessive leakage found in the model tests. The writers feel that the remaining $3^1/_2$ per cent is much too great a step-up to be warranted with the present knowledge.

Figs. 15 and 16 of this discussion show the comparative performances of the double-volute pump, the fixed-vane diffuser pump, and the adjustable wicket-gate pump. Fig. 15 is plotted on the basis of 150 rpm prototype operation, while Fig. 16 is calculated for the same machines but operating at a speed corresponding to 180 rpm prototype operation. Capacities and heads are represented in percentage of normal in accordance with the convention used in the previous paper (12). However, actual efficiencies of the test pumps are plotted in the place of relative efficiencies formerly used. The first point that is observed

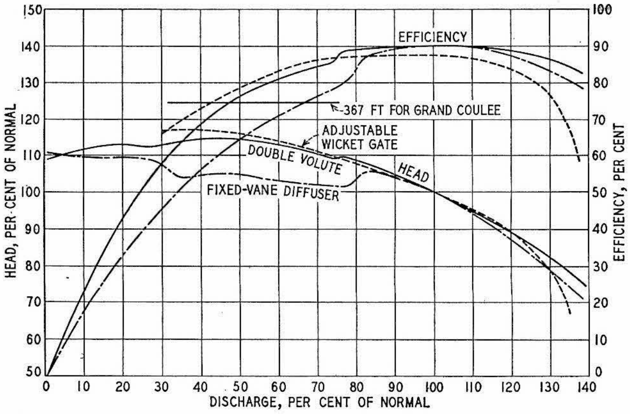

FIG. 15 COMPARISON OF CASING EFFECTS ON CHARACTERISTICS; DOUBLE-VOLUTE, FIXED-VANE DIFFUSER, AND ADJUSTABLE WICKET-GATE PUMPS

(Prototype speed, 150 rpm.)

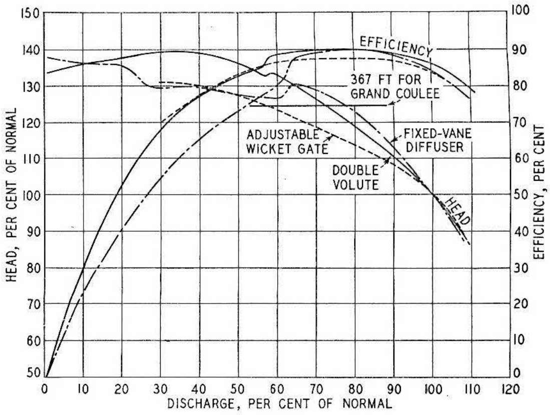

FIG. 16 COMPARISON OF CASING EFFECTS ON CHARACTERISTICS; DOUBLE-VOLUTE, FIXED-VANE DIFFUSER, AND ADJUSTABLE WICKET-GATE PUMPS

(Prototype speed, 180 rpm.)

in Fig. 15 is that none of the pumps when operated at the design speed would develop sufficient head to deliver against the maximum head encountered in the Grand Coulee installation. The second point of interest is that, over a surprisingly wide range, the double-volute and the wicket-gate pumps have nearly identical capacity-head curves. A comparison of the efficiency curves shows that both the double-volute and the fixed-vane diffuser pumps have maximum efficiencies about 3 per cent higher than the wicket-gate machine. Of this difference, 1 per cent is undoubtedly due to the excessive leakage loss previously mentioned, but the remaining 2 per cent is probably chargeable to the additional amount of friction surface in the high velocity flow.

The broad range of high-efficiency operation of both the wicket-gate and double-volute pumps is very apparent. Calculation shows that a range of capacity of approximately 61 per cent of the normal is covered with an efficiency within 5 per cent of the maximum by the wicket-gate machine, whereas, the corresponding value for the double-volute is about 58 per cent and for the fixed-vane diffuser about 43 per cent. The adjustable wicket-gate pump has a noticeably higher efficiency for all discharges below 60 per cent of the normal.

If Fig. 16 is now examined, it will be seen that the situation is changed. In the first place, all three pumps now are able to meet the required range of head when operated at 180 rpm. However, the relative steepness of the head curves is no longer the same. The fixed-vane diffuser case shows the best performance, giving a delivery of 78 per cent of normal capacity at the maximum head. The double-volute pump is next with 73 per cent, whereas, the adjustable wicket-gate pump delivers only 58 per cent. At this higher operating speed, the double-volute case shows a considerably wider range of high-efficiency operation than does either of the other two. The adjustable wicket-gate unit still shows a 3 per cent lower maximum efficiency and, in addition, no longer shows a better efficiency for the lower capacities. These conclusions are somewhat surprising when it is remembered that the curve shown here for the wicket-gate pump is the envelope for the different gate positions.

In the concluding paragraph of the paper, the authors give seven advantages of the wicket-gate construction over the non-adjustable cases. Figs. 15 and 16, of this discussion, offer a concrete means of evaluating these items as follows:

1 While the power curves for the three cases are not shown in either of these figures, it is recognized that the shutoff power for the wicket-gate pump is definitely lower, provided,of course, that the flow is controlled by closing the wicket gates themselves.

2, 3 These depend upon the speed selected for making the comparison. As was shown in the discussion of Figs. 15 and 16, the difference is small for the 150-rpm operation, but for the 180-rpm operation the double-volute case shows a flatter efficiency discharge curve, and both it and the fixed-vane diffuser case show steeper head curves.

4 Neither Fig. 15 nor Fig. 16 shows an appreciable increase in discharge-overload capacity in the lower heads. This is surprising for the 180-rpm operation, because of the flatter head curve of the wicket-gate pump. Here the lack of overload capacity is probably due, as the authors suggest, to cavitation.

5 It is certainly true that wicket gates form an excellent means of controlling discharge at constant speed and head.

6 The adjustable wicket-gate pump had a smaller impeller diameter when compared on a prototype basis. For the three pumps shown in Figs. 15 and 16, the adjustable wicket-gate pump would have an impeller diameter of 16.7 ft, the double-volute pump 16.95 ft, and the fixed-vane diffuser 17.1 ft. For this particular group, this represents a range of 2.4 per cent. However, this is not the entire story. The figure which best characterizes the casing sizes is the one given by the authors in Table 1, as "offset of casing throat to center line." Again, on the basis of 150 rpm prototype speed, this distance would be $15^1/_4$ ft for the adjustable wicket-gate pump, $13^1/_2$ ft for the double-volute pump, and $14^1/_2$ ft for the fixed-vane diffuser. Thus it is seen that, while the adjustable wicket-gate machine has the smallest impeller diameter, it has the greatest over-all case diameter. The cross-sectional areas of the spiral case surrounding the wicket gates and stay vanes will be larger also since the effect of the adjustable wicket gates is to reduce case velocities.

7 In units of this size, the structural design of a single-volute case is quite difficult. The introduction of a second volute overcomes a large part of this disadvantage. On the other hand, the stay vanes used in both the fixed-vane-diffuser and the adjustable wicket-gate pumps make possible a very satisfactory structural design.

In conclusion, the writers would like to make a few comments regarding the operation of these units as turbines. In the light of the comparison of the pump characteristics, it would appear that the chief advantage to be gained by the use of the adjustable wicket-gate case lies in the region of turbine operation and that, therefore, the whole discussion must be on a basis of the combined operation both as pump and as turbine. If, when the unit is operated as a turbine, the system conditions are such that it must be governor-controlled, then the wicket-gate construction offers advantages which cannot be obtained with the other cases. However, if the system conditions permit the utilization of the unit to supply a block load, then governor-controlled operation is not necessary. For this condition, any of the case types gives satisfactory operation. For the three units under comparison, both the fixed-vane diffuser and the double-volute type show higher efficiencies as turbines than does the adjustable wicket gate. The wicket gate, however, does have the one advantage in that a wider range of power output can be obtained for a given head for the same relative variation of efficiency. For the non-adjustable type case, unless a throttle valve is incorporated in the system, there is only one output. For a multiple-unit installation this seems to offer no disadvantage.

It should be emphasized that the writers have been as objective as possible in making comparisons, limiting statements to the results which were substantiated by the tests. The laboratory staff feels that each of the different casing types has its own field of application and, therefore, few generalizations are justified. The present comparisons have as a background the Grand Coulee requirements. Other installations, which have entirely different operating conditions, would result in a completely different order of desirability of casing requirements.

R. E. B. Sharp.[13] Discussions of the performance of pump-turbines are in order, in view of the fact that the present capacity demand has recently awakened decided interest in pumped-storage projects.

For comparison with the authors' tests, Baldwin-Southwark Model Runner 115, tested during July, 1931, and Model Runner 117, tested during July, 1933, may be of interest.[14] Model 115 was designed for peak-load storage plants with relatively high heads, and Model 117 for medium heads. The specific speed of Model 115, when pumping, at best efficiency, is 1970, as compared to the authors' 1750; and when generating is 26, as compared to the authors' 25. Runner 117 has a specific speed of 2700 when pumping and 41 when generating. These tests were made under low-head conditions (3 to 4 ft).

While the resulting Reynolds number is low, it has been demonstrated as being sufficiently high to be well within the turbulent region of flow and to form a reliable comparison with prototype performance and with model tests under higher heads. The discharge was measured by a calibrated weir, and the power by electric dynamometer. The head, acting both for pumping and generating, was considered as the vertical difference between headwater and tailwater levels. The tests were made with vertical shafts, with the volute casing submerged, and with draft elbows of usual turbine design. The draft-tube losses are charged against the pump-turbine for both cycles of operation.

Figs. 17 and 18 of this discussion show characteristics, in general, similar to the authors'. The same tendency for irregularity in the curves to the left of the maximum-efficiency point was encountered.

The practical utilization of pump-turbines, in virtually all cases, requires that the head when pumping be somewhat greater than when generating, due to the friction losses both ways in the penstock. The pumping operation, although occurring at times of low system load, requires for its justification the minimum expenditure of energy. Therefore, both the pumping and generating cycle should be at the most efficient condition.

In Fig. 17, the performance of model 115 as a pump has been stepped up to 1000 rpm, the head at best efficiency being 111 ft. In Fig. 19, the performance of this model as a turbine has been plotted when operating at the same head of 111 ft. The curves for two speeds are shown, the lower efficiency curve being for 1000 rpm, and the higher efficiency and output for a speed of 868 rpm. It will be noted that the loss in efficiency for uniform speed of operation during both cycles is about 5 per cent. Therefore, in practice, two speeds are necessary, the indicated loss with uniform speeds being aggravated by the pumping head being in excess of the generating head. As an alternative, a booster pump in series with the pump-turbine can be utilized to reduce the pumping head on the pump-turbine so that maximum efficiency will be attained at a uniform speed.

In Fig. 18 of this discussion, the performance of model 117, as a pump, has been stepped to 1000 rpm, the head at best efficiency being 86.75 ft. In Fig. 20, the performance of this model as a turbine has been plotted, when operating at the same head (86.75 ft). In this case also, the curves for two speeds are shown, the lower one being for 1000 rpm; the higher curve is for 900 rpm. It is interesting to note that, for this higher-specific-speed

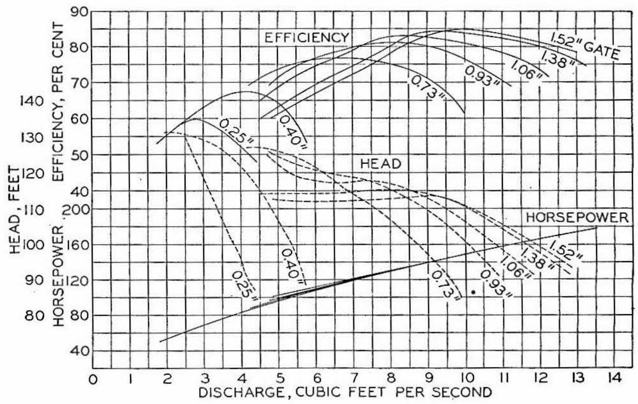

Fig. 17 Baldwin-Southwark Runner No. 115, Operating as a Pump

(Diameter of runner, 18 in.; speed, 1000 rpm.)

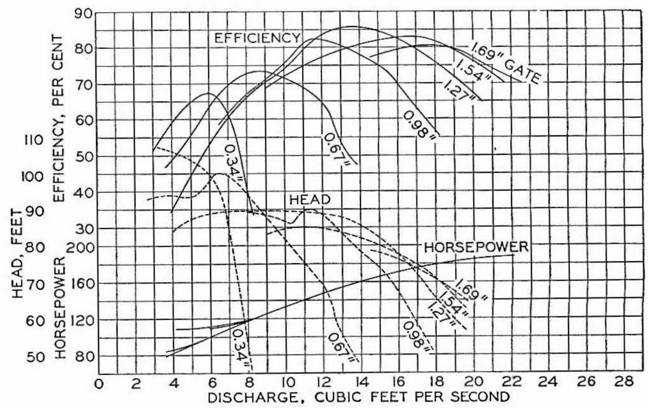

Fig. 18 Baldwin-Southwark Runner No. 117, Operating as a Pump

(Diameter of runner $16^5/_{16}$ in.; speed 1000 rpm.)

[13] Chief Engineer, I. P. Morris Department, Baldwin-Southwark Division, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

[14] Note the following pertinent patents: No. 1,494,008, issued May 13, 1924, Method and Means for Converting Energy, by Forrest Nagler, assigned to Allis-Chalmers Manufacturing Company; No. 1,919,376, Reversible Pump-Turbine, by L. F. Moody; No. 1,941,-361, Air Inlet Control and Method of Operating a Pump-Turbine, by L. F. Moody; No. 2,010,555, Hydraulically Reversible Pump-Turbine, by L. F. Moody; No. 2,246,472, Hydraulic Power Accumulation System, by R. E. B. Sharp, assigned to Baldwin Locomotive Works.